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NASA Technical Memorandum 106848

Wave Journal Bearing
Part II: Experimental Pressure Measurements and Fractional Frequency Whirl Threshold for Wave and Plain Journal Bearings

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Prepared for the
Energy and Environmental Expo 95
sponsored by the American Society of Mechanical Engineers
Houston, Texas, January 29—February 1, 1995

National Aeronautics and
Space Administration

(NASA-TM-106848-Pt-2) WAVE JOURNAL BEARING. PART 2: EXPERIMENTAL PRESSURE MEASUREMENTS AND FRACTIONAL FREQUENCY WHIRL THRESHOLD FOR WAVE AND PLAIN JOURNAL BEARINGS (NASA. Lewis Research Center) 10 p

N95-22405

Unclas

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WAVE JOURNAL BEARING; PART II: EXPERIMENTAL PRESSURE MEASUREMENTS AND FRACTIONAL FREQUENCY WHIRL THRESHOLD FOR WAVE AND PLAIN JOURNAL BEARINGS

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ABSTRACT

A new hydrodynamic bearing concept, the wave journal bearing, is being developed because it has better stability characteristics than plain journal bearings while maintaining similar load capacity. An analysis code to predict the steady state and dynamic performance of the wave journal bearing is also part of the development. To verify numerical predictions and contrast the wave journal bearing's stability characteristics to a plain journal bearing, tests were conducted at NASA Lewis Research Center using an air bearing test rig. Bearing film pressures were measured at 16 ports located around the bearing circumference at the middle of the bearing length. The pressure measurements for both a plain journal bearing and a wave journal bearing compared favorably with numerical predictions. Both bearings were tested with no radial load to determine the speed threshold for self-excited fractional frequency whirl. The plain journal bearing started to whirl immediately upon shaft start-up. The wave journal bearing did not incur self-excited whirl until 800 to 900 rpm as predicted by the analysis. Furthermore, the wave bearing's geometry limited the whirl orbit to less than the bearing's clearance. In contrast, the plain journal bearing did not limit the whirl orbit, causing it to rub.

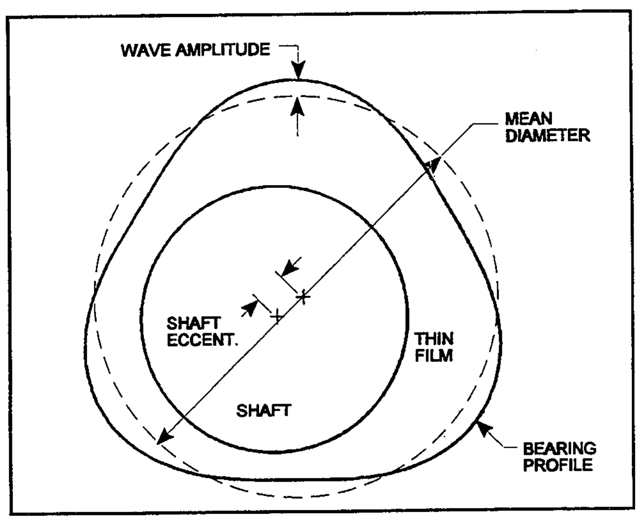


FIGURE 1. THREE-WAVE JOURNAL BEARING (WAVE AMPLITUDE AND CLEARANCE GREATLY EXAGGERATED)

INTRODUCTION

The stability characteristic of fluid film bearings is critical for successful application to rotating machinery. Of prime concern is self-excited fractional frequency whirl which can quickly lead to bearing and rotating machinery failure. Self-excited whirl is caused by the cross-coupling stiffness inherent in the hydrodynamic effects of fluid film bearings. Commonly used plain journal bearings are particularly sensitive to this type of instability because their theoretical load capacity becomes zero if the shaft whirls about the center of the bearing (1,2,3). The plain journal bearing's stability can be improved by modifying its geometry. Numerous concepts including lobed, grooved, stepped, and tilting-pad bearings have been developed (4,5). These concepts, however, have substantially lower load capacity than the plain journal bearing.

The wave journal bearing concept, which is displayed in figure 1, is being developed at the NASA Lewis Research Center to improve the stability characteristics over plain journal bearings while maintaining similar load capacity (6,7). An analysis code to predict steady state and dynamic performance of the wave journal bearing is part of the development. The code can model plain journal bearings and wave journal bearings with any number of waves and different wave amplitudes (8,9). To validate the analysis, an air bearing test rig was built at the NASA Lewis Research Center. Preliminary test results of a three-wave and a plain journal bearing were presented in references 10 and 11. The results reported in these references included film pressure measurements for the two bearing types subjected to various radial loads and at speeds ranging from 3,000 to 10,000 rpm. Both references reported good agreement between predictions and experimental data.

To validate numerical predictions of the bearing's threshold for self-excited whirl, additional tests of plain and wave journal bearings were conducted in the NASA air bearing test rig. The speed threshold for self-excited whirl and its characteristics were determined with no radial load applied to the bearings. In addition, film pressure data were obtained for the plain journal bearing subjected to several different radial loads and for the wave journal bearing with no radial load to provide insight into the wave journal bearing's enhanced stability. The results are compared to numerical predictions.

EXPERIMENTAL METHODS

An air bearing test rig (figure 2) was built to test wave journal bearings with 50 mm (2.0 inch) diameter and 58 mm (2.3 inch) length. A commercial air spindle with a removable (add-on) shaft is used to drive the test bearing. It is mounted vertically to eliminate gravitational effects. The spindle can operate to 30,000 rpm with a shaft run-out of less

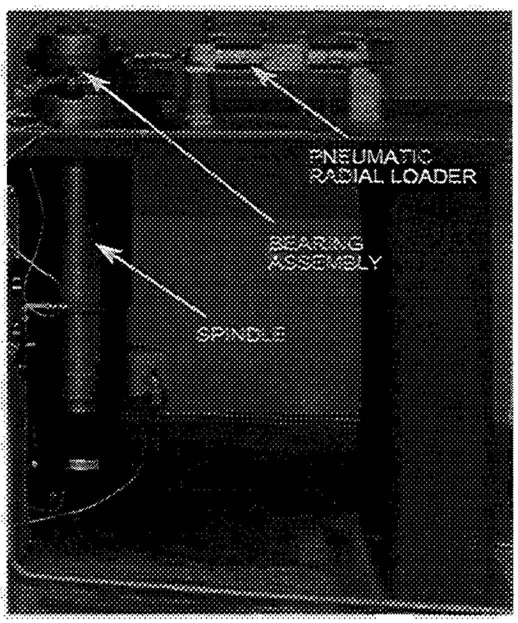


FIGURE 2. BEARING TEST RIG

than 1 micron (40 μ-inches). The mechanical run-out of the add-on shaft was measured to be less than 1 micron (40 μ-inches). A pneumatic load cylinder can be used to apply a radial load to the bearing.

The test bearing is levitated between two thrust plates (figure 3) using pressurized air. The thrust plates enable the test bearing to translate in the radial direction and also to rotate about the vertical-axis. The bottom thrust plate was

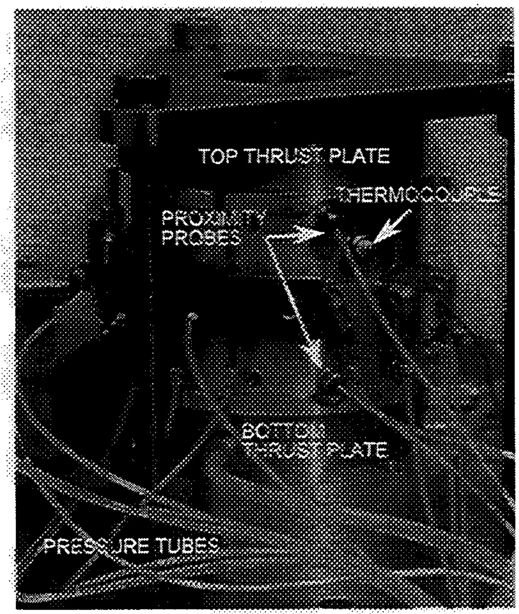


FIGURE 3. BEARING ASSEMBLY

designed with three thrust pads to allow bearing alignment to the shaft. The top thrust plate contains a single 360 degree pad. It is supported by three threaded rods which permit adjustment to the axial clearance of both thrust plates. For safety considerations, a rod that protrudes radially from the bearing housing triggers a micro-switch that stops the spindle if the shaft exerts excessive torque on the bearing housing. The motion of the rod was useful for determining if the whirl orbits caused the shaft to rub the bearing.

The test bearings were operated in ambient air. However, the bearing end plane pressure closest to the spindle was slightly higher than atmospheric pressure because it was bounded by the spindle and thrust plate, restricting the flow of air from the area.

Instrumentation

The test rig includes instrumentation to measure shaft speed, radial load, shaft orbit, bearing temperature, and bearing film pressure. Shaft speed is measured by a capacitive probe located within the spindle. A miniature precision load cell measures the radial load applied by a pneumatic load cylinder. Four inductive displacement sensors allow measurement of shaft orbits at each end of the bearing. The bearing housing temperature is monitored with a thermocouple. Bearing steady state film pressures are measured at 16 ports located around the circumference in the middle plane of the bearing (figure 4). Strain-gage type transducers are used to measure absolute pressures. In addition, both bearing end pressures are measured.

Test Specimens

A plain journal bearing and a three wave journal bearing were tested on a chrome plated shaft with a 50.891 mm (2.0036 inches) diameter. Both bearings were made of

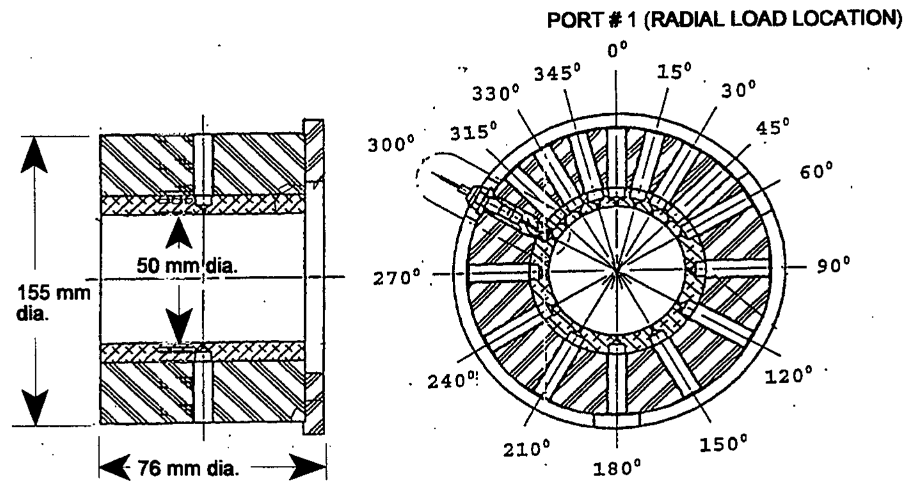


FIGURE 4. PRESSURE PORT LOCATIONS (MIDDLE OF BEARING LENGTH)

anodized aluminum to resist damage caused by potential rubs. The plain journal bearing deviated only slightly from a perfectly circular bearing as indicated by its small 0.034 distortion ratio, which is defined as the radial distortion from perfectly circular divided by the bearing clearance. The diameter of the plain journal bearing is 50.963 mm (2.0064 inches) and the length is 58.014 mm (2.284 inches). The radial clearance is 36 µm (1.4 mils).

The measured profile of the wave journal bearing is shown in figure 5. The bearing has a mean diameter of 50.965 mm (2.0065 inches) and a wave amplitude of 13 µm (0.5 mils). Thus, the mean radial clearance and wave amplitude ratio are 37 µm (1.5 mils) and 0.35, respectively.

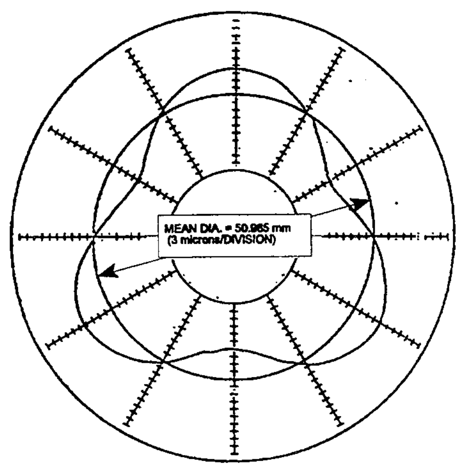


FIGURE 5. MEASURED PROFILE FOR THE THREE-WAVE BEARING

Test Procedures

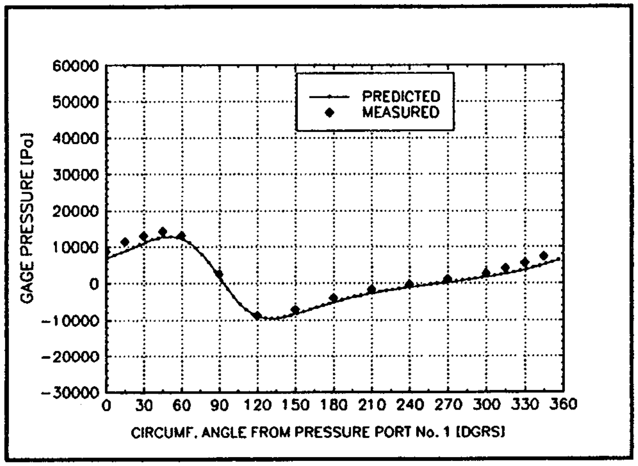
Stability tests were conducted without the pressure tubes connected to the bearing housing and the ports capped to eliminate damping effects caused by the plastic tubing. The wave bearing mass was varied by adding portions of a ring (shown in figure 12), used as part of the radial load system, to determine the effects on bearing stability. Testing proceeded by manually increasing the shaft speed until fractional frequency whirl was experienced. The process was repeated for each bearing to determine the repeatability of the onset.

RESULTS AND DISCUSSION

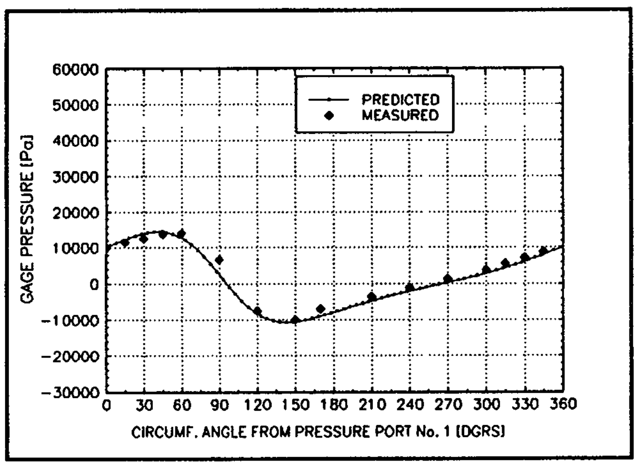
The plain and wave journal bearings were tested to investigate stability characteristics and obtain the circumferential bearing film pressure distribution. The data are presented and compared to theoretical predictions.

Plain Journal Bearing Tests

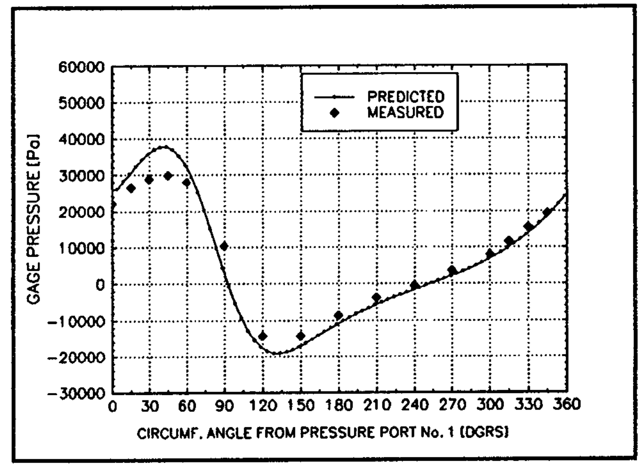
The plain journal bearing was tested to 20,000 rpm with various radial loads. A radial load was applied and gradually increased with speed to prevent self-excited fractional frequency whirl. The predicted pressures are plotted with the experimental data in figure 6 for four speed and load combinations. The predictions agree well with the data. A changing bearing clearance caused by a rising bearing temperature was taken into account in the predicted pressures. As expected, the plain journal bearing generated a single pressure wedge. The peak of the pressure wedge is not directly opposite the applied radial load. Furthermore, the resulting reaction film pressure force is offset from the direction of the applied radial load, and therefore a component of the reaction



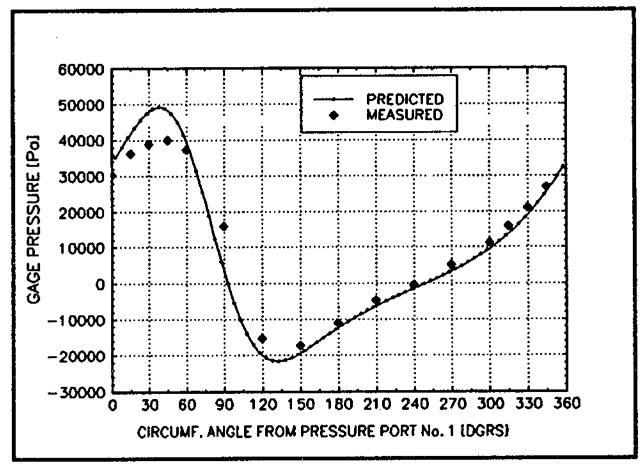
a. 5,100 RPM, RADIAL LOAD = 25 N,
RADIAL CLEARANCE = 38 MICRONS



b. 10,160 RPM, RADIAL LOAD = 33 N,
RADIAL CLEARANCE = 38 MICRONS



c. 15,000 RPM, RADIAL LOAD = 69 N,
RADIAL CLEARANCE = 38 MICRONS



d. 20,000 RPM, RADIAL LOAD = 88 N,
RADIAL CLEARANCE = 39 MICRONS

FIGURE 6. BEARING FILM PRESSURE AT THE MIDDLE SECTION FOR A PLAIN JOURNAL BEARING

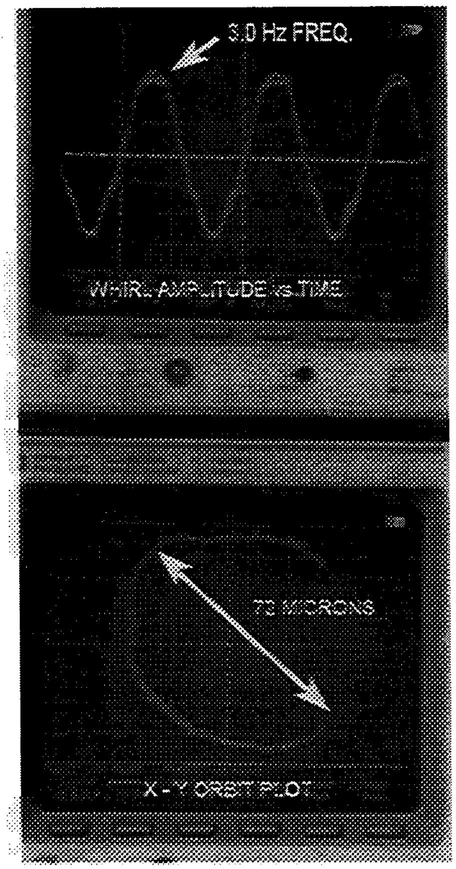


FIGURE 7. HALF FREQUENCY WHIRL MOVEMENT OF A PLAIN JOURNAL BEARING AT 360 RPM SPEED

force is perpendicular to the applied load. If the perpendicular component becomes large enough, the bearing will start to whirl. This inherent property of the hydrodynamic bearing is commonly referred to as the cross coupling effect.

Without a radial load, the bearing became unstable immediately upon shaft start-up. Figure 7 displays a large subsynchronous whirl occurring at 360 rpm. As expected, the plain journal bearing whirled at half the shaft synchronous frequency. The bearing translated in a whirl motion as indicated by figure 7, (top picture) which displays the output of two proximity probes located at the same angular position but at two different axial locations. The two signals appear as one because they have the same magnitude and no phase difference; evidence that the bearing remained parallel to the shaft. The whirl orbit is the same size as the clearance indicating that the shaft is contacting the bearing. The rod used to sense excessive bearing torque remained pressed up against the micro-switch, corroborating that the shaft was rubbing the bearing; the resulting torque, however, was insufficient to shut down the spindle (see figure 8). In contrast, the rod floated freely when the bearing ran free of fractional frequency whirl. In ref 7, the threshold was reported to occur at a much higher speed, but it is suspected that

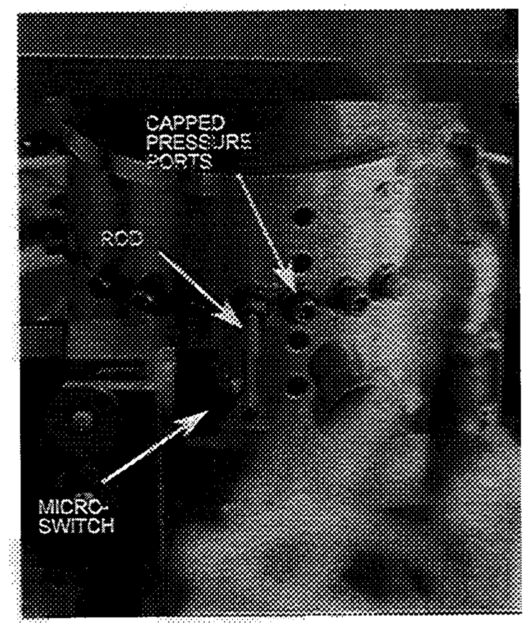


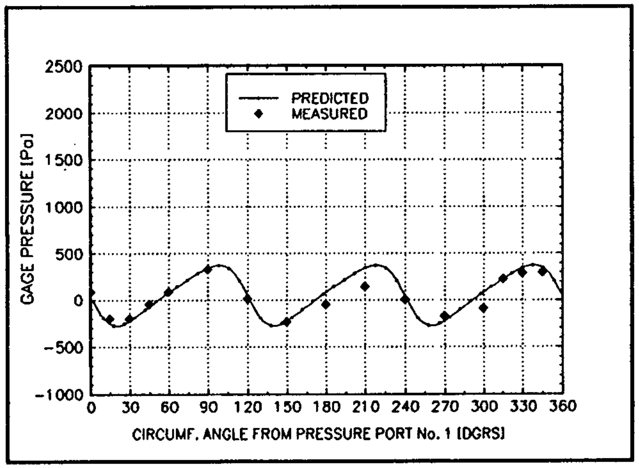
FIGURE 8. ROD POSITION FOR THE PLAIN JOURNAL BEARING EXPERIENCING WHIRL

this was caused by the added damping provided by the pressure tubes that were connected to the bearing housing.

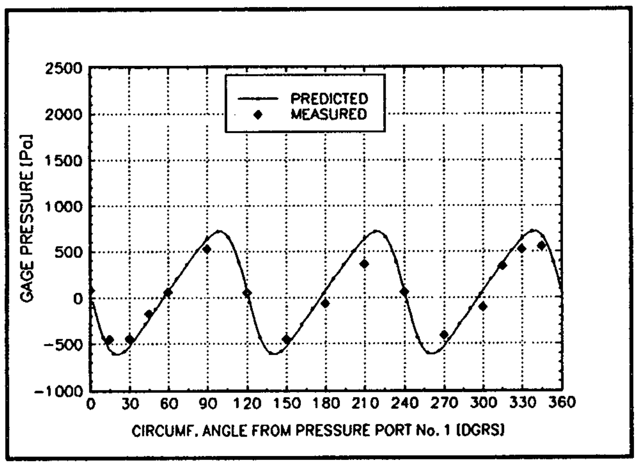
Three Wave Bearing Tests

The three-wave bearing was tested at speeds up to 2,500 rpm without a radial load and free of fractional frequency whirl to measure the circumferential film pressure distribution. The experimental data and numerical predictions for four different speeds are displayed in figure 9. Predicted pressures show good agreement with the experimental data. In contrast to the plain journal bearing, the wave bearing generates pressure wedges even when it is unloaded. Three pressure wedges were generated evenly spaced around the circumference of the bearing. The generated pressure restrains bearing whirl or any deviation from the concentric position, enabling stable operation to high speeds. The bearing was then tested with the plastic pressure tubes removed from the housing. Without a radial load, a fractional frequency whirl orbit appeared at a speed of approximately 900 rpm. On several tests the threshold speed varied from 800 to 900 rpm. The orbit, which is shown in figure 10, was approximately half the bearing clearance. The threshold speed was higher when the pressure tubes were connected to the bearing housing because the tubes added damping. The fractional frequency whirl orbit grew to about 70% of the bearing clearance at 1,320 rpm as shown in figure 11, and it remained at this amplitude to 2,500 rpm. The bearing limits the whirl orbit to less than the clearance as evidenced by the fact that the rod that was used to sense excessive torque floated freely throughout the tests (see figure 12).

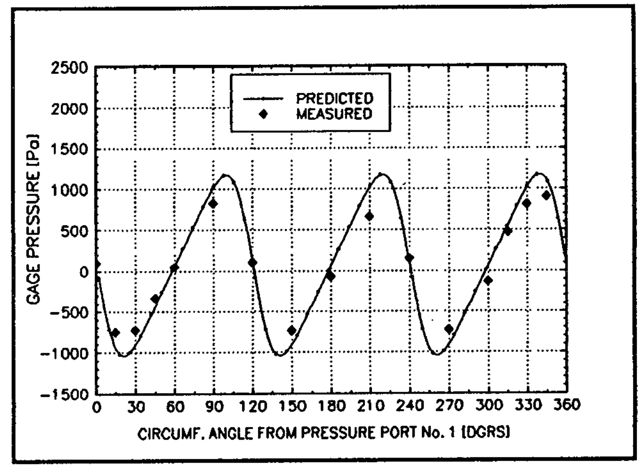
Data from the proximity probe provides further insight into the enhanced stability behavior of the wave bearing. The top picture of figure 11, showing the whirl path as a function of



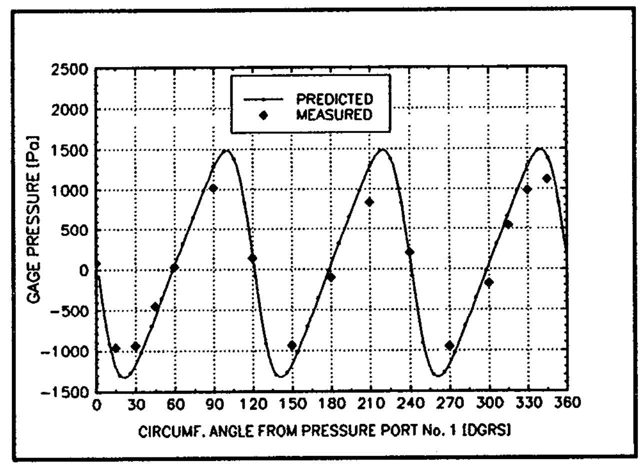
a. 600 RPM, RADIAL CLEARANCE = 39 MICRONS AND WAVE AMPLITUDE RATIO = .33 AT BEARING TEMPERATURE = 29.5 C



b. 1,200 RPM, RADIAL CLEARANCE = 39 MICRONS AND WAVE AMPLITUDE RATIO = .33 AT BEARING TEMPERATURE = 29 C



c. 2,000 RPM, RADIAL CLEARANCE = 39 MICRONS, WAVE AMPLITUDE RATIO = .33 AT BEARING TEMPERATURE = 29 C



d. 2,500 RPM, RADIAL CLEARANCE = 39 MICRONS AND WAVE AMPLITUDE RATIO = .33 AT BEARING TEMPERATURE = 28 C

FIGURE 9. BEARING FILM PRESSURE AT THE MIDDLE SECTION FOR A THREE WAVE JOURNAL BEARING WITH NO RADIAL LOAD

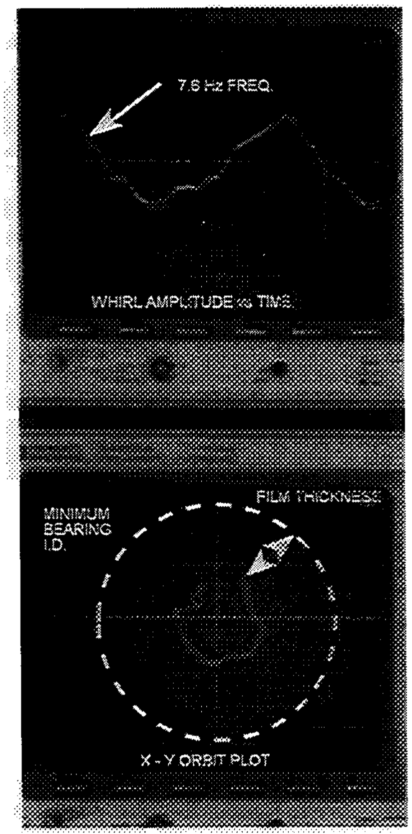


FIGURE10. FRACTIONAL FREQUENCY WHIRL MOVEMENT OF A THREE-WAVE BEARING AT 920 RPM SPEED

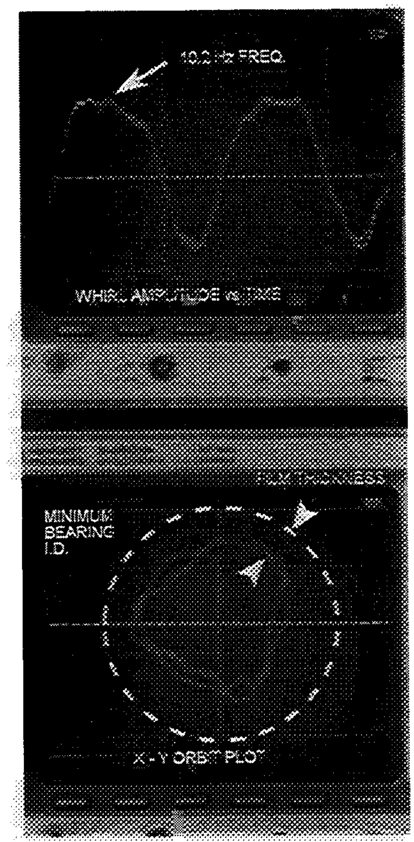


FIGURE 11. FRACTIONAL FREQUENCY WHIRL MOVEMENT OF A THREE-WAVE BEARING AT 1,320 RPM SPEED

time, displays a non-sinusoidal curve. The one peak is flattened, indicating the whirl orbit is restrained. The x-y plot shown in the bottom picture is more informative showing the whirl orbit collapsed inward at three locations. This is caused by the pressure wedges generated by the wave geometry. The pressure wedges restrain the growth of the self-excited bearing whirl. The same orbit shape is not present at the lower speeds because the hydrodynamic forces are smaller (see figure 10).

Bearing critical mass was also experimentally determined to validate the predictions for self-excited whirl. Since the threshold speed varies as a function of bearing mass, several tests were conducted using different bearing masses. Figure 13 displays the experimental data plotted on the prediction curves. There are two curves because the bearing temperature varied between the tests. The analysis predicts that the speed threshold for fractional frequency whirl will decrease as the bearing mass increases, and the experimental data substantiated the predicted trend. Furthermore, the good agreement between the predictions and experimental data provides strong evidence that the dynamic analysis is valid since the critical mass is calculated from the primary outputs of the dynamic analysis.

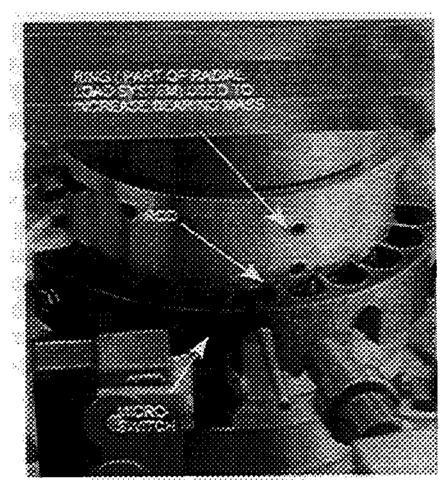


FIGURE 12. ROD POSITION FOR THE THREE- WAVE BEARING EXPERIENCING WHIRL MOTION

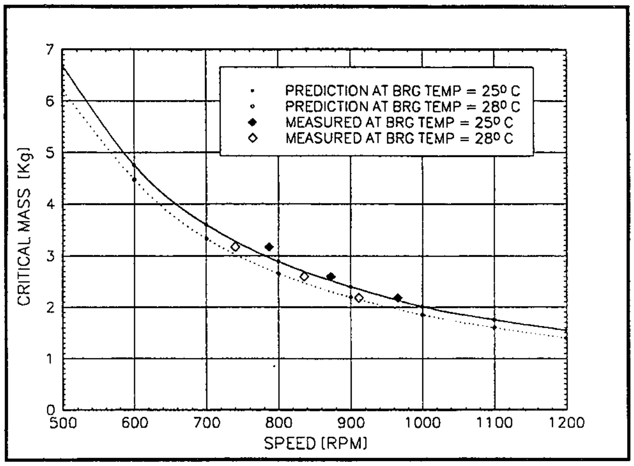


FIGURE 13. PREDICTED AND EXPERIMENTAL CRITICAL MASS FOR A THREE WAVE JOURNAL BEARING WITH NO RADIAL LOAD. (At Brg temp=25C:Radial Clearance=37.8microns, Wave Ampl=.338; At Brg temp=28C, Radial Clearance=38.9 microns, Wave Ampl=.330)

CONCLUDING REMARKS

A plain journal bearing and a three-wave journal bearing were tested in an air bearing test rig to verify pressure predictions and contrast stability characteristics. The numerical predictions of film pressure show good agreement with experimental data. The results revealed significant differences between the pressure profiles for the two bearings.

While the plain journal bearing did not generate pressure when tested without a radial load, the wave journal bearing generated three pressure wedges that were equally spaced around the bearing's circumference. The generated pressure is believed to be the basis for the enhanced stability of the wave bearing. The plain journal bearing is very susceptible to the fractional frequency whirl movement when operated without a radial load. It becomes unstable immediately upon shaft rotation. Furthermore, the bearing is unable to limit the fractional frequency whirl orbit. The orbit grows until the bearing and shaft rub. This type of bearing requires a radial load to remain whirl free and stable.

In contrast, the unloaded three-wave journal bearing remained stable and free of fractional frequency whirl up to about 900 rpm. The bearing continued operating even with fractional frequency whirl motion as the speed was increased to 2,500 rpm. The wave bearing's geometry enables it to limit the whirl orbit to less than its clearance. In addition, predictions of bearing critical mass were verified with experimental data, establishing good evidence that the dynamic analysis is valid.

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REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	January 1995	Technical Memorandum

4. TITLE AND SUBTITLE

Wave Journal Bearing
Part II: Experimental Pressure Measurements and Fractional Frequency Whirl Threshold for Wave and Plain Journal Bearings

5. FUNDING NUMBERS

WU–778–32–21

6. AUTHOR(S)

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7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135–3191

8. PERFORMING ORGANIZATION REPORT NUMBER

E–9424

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Washington, D.C. 20546–0001

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

NASA TM–106848

11. SUPPLEMENTARY NOTES

Prepared for the Energy and Environmental Expo 95 sponsored by the American Society of Mechanical Engineers, Petroleum Division, Houston, Texas, January 29—February 1, 1995. James F. Walker and Harold E. Addy, Jr., NASA Lewis Research Center; Florin Dimofte, University of Toledo, Toledo, Ohio 43606 (work funded by NASA Grant NAG3–1370). Responsible person, James F. Walker, organization code 5310, (216) 977–7465.

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Unclassified - Unlimited
Subject Categories 18, 20, 07, and 37

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14. SUBJECT TERMS

Bearings; Journal bearings; Stability tests

15. NUMBER OF PAGES

10

16. PRICE CODE

A02

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	